Arun Field High-Pressure Gas Reinjection Facilities

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Summary

This paper discusses the high-pressure natural gas reinjection facilities at the Mobil Oil Arun natural gas field in north Sumatra. Each of the three compression trains at Arun comprises a four-stage centrifugal compressor with a 28,000-bhp [20 888-kW] gas turbine driver plus associated cooling, scrubbing, and control facilities. The present throughput of each train is around 250×10^6 scf/D [7.08×10^6 std m^3/d] at 6,100 psig [42 MPa] at 6,500-psig [44-MPa] discharge pressure. The paper first discusses the overall Arun field facilities and operation. It then describes in detail the compression train facilities including the causes of, and solutions to, the many and various problems associated with the commissioning and operation of these prototype compression facilities. The paper also describes performance monitoring of the trains and the efforts made to optimize train capacity. The paper concludes with 12 specific practical recommendations based on the problems and experiences encountered at Arun during the last 7 years' operation.

Arun Field Development

Field Statistics. Arun field is a giant natural gas field situated onshore north Sumatra in the province of Aceh.

The natural gas reservoir is a carbonate reef at a depth of 10,000 ft [3048 m]. Initial reservoir pressure was 7,100 psig [49 MPa].

The field, discovered in 1971, became operational in 1977. The current field production is $2,300 \times 10^6$ scf/D [65.13×10^6 std m^3/d] natural gas with an associated liquid condensate production of 140,000 B/D [22 258 m^3/d]. Of the gas, $1,550 \times 10^6$ scf/D [43.89×10^6 std m^3/d] goes to the P.T. Arun liquefied natural gas (LNG) plant, which is 19 miles [30 km] away on the northeast coast of Sumatra. The remaining gas production, around 750×10^6 scf/D [21.24 std m^3/d], is reinjected into the Arun reservoir by using three gas-turbine-driven centrifugal compressors. The reinjection facilities act primarily as a backup to ensure a continuous gas flow to the LNG plant in the event of a production upset. Moreover, the additional gas production has accelerated the production of liquid condensate and, hence, field revenue.

The Cluster Concept. The Arun field production facility is designed around a "cluster" concept. At present there are four clusters sited along the length of the field, with 16 production wells per cluster. Each cluster is an independent, self-contained production area except for electrical power, which is generated at a central common facilities area. Each cluster has two identical production trains (Fig. 1) designed to handle 375×10^6 scf/D [10.62×10^6 std m^3/d] dry gas. Each train has independent pressure-reduction, cooling, liquid separation, and condensate-pumping facilities.

Reinjection Facilities. The four clusters are essentially identical to each other except for Clusters 2 and 3, which also site the three Arun field gas reinjection trains. There are two compressor trains at Cluster 3 (K301 and K351), and one compressor train at Cluster 2 (K201).

The suction to the compressors is taken from the cluster gas production pipeline and the discharge goes to the cluster gas injection header (Fig. 2). From the header, separate lines transport the injection gas to each of nine gas injection wells (GIW's). Six GIW's tie in to the Cluster 3 injection header and three GIW's into the Cluster 2 injection header. There is also an injection header interlink line between Clusters 2 and 3 for balancing of GIW flows if required.

Gas Reinjection Trains

Design Parameters. The three identical compressor trains at Arun *originally* were designed to the conditions listed in Table 1.

Description of Main Compressors. The original injection compressors were skid-mounted, four-stage, centrifugal compressors, consisting of a 272B four to four low-pressure barrel (Fig. 3) and a 181B three to three (Fig. 4) high-pressure barrel.

The compressors comprise the following main components.

Compressor Case. Both the 272B and 181B compressor sections comprise a one-piece, forged steel barrel containing a bundle and rotor assembly. The case is the main pressure-containing device of the compressors. The 272B case is pressure tested to 7,500 psig [52 MPa] and the 181B case to 12,750 psig [88 MPa].

Bundle Assembly. The bundles include the intake walls, inlet guide vanes, diaphragms, and division walls. They are horizontally split with O-ring cordage for sealing between halves and taper pins for assembly alignment. O-rings also are used for interstage sealing between the bundle and the compressor case.

Rotor Assemblies. These consist of a shaft, impellers, (four per stage on the 272B shaft and three per stage on the 181B shaft), impeller spacers, and thrust-bearing disks. The impellers of each stage are mounted in a back-to-back configuration to reduce the axial thrust loads

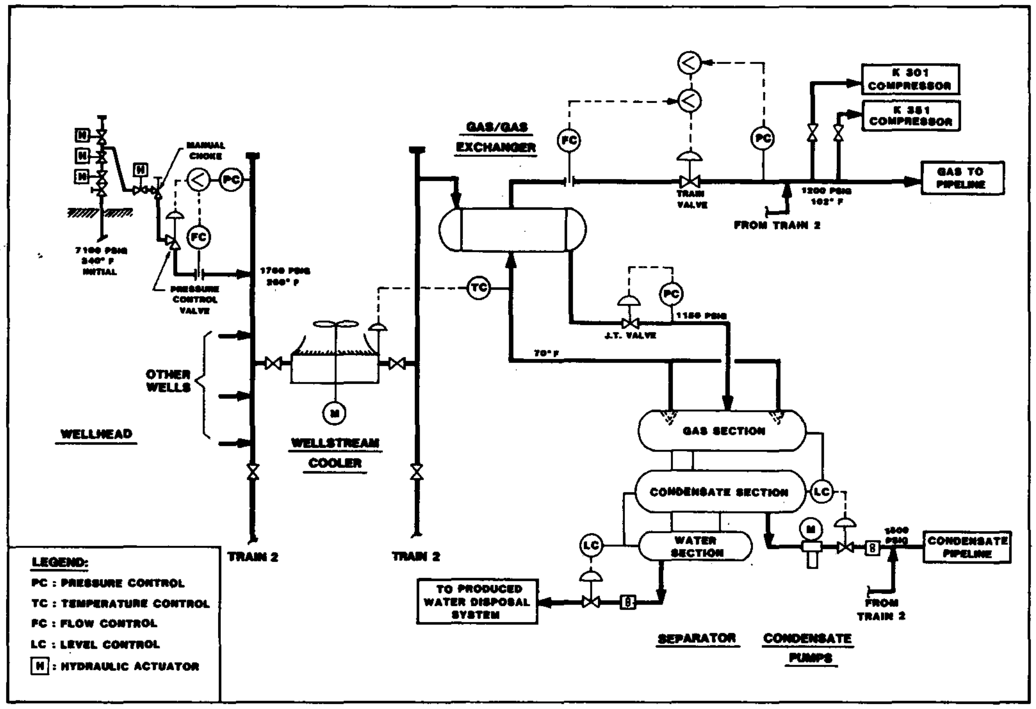

Fig. 1—Production train flow diagram (Cluster 3).

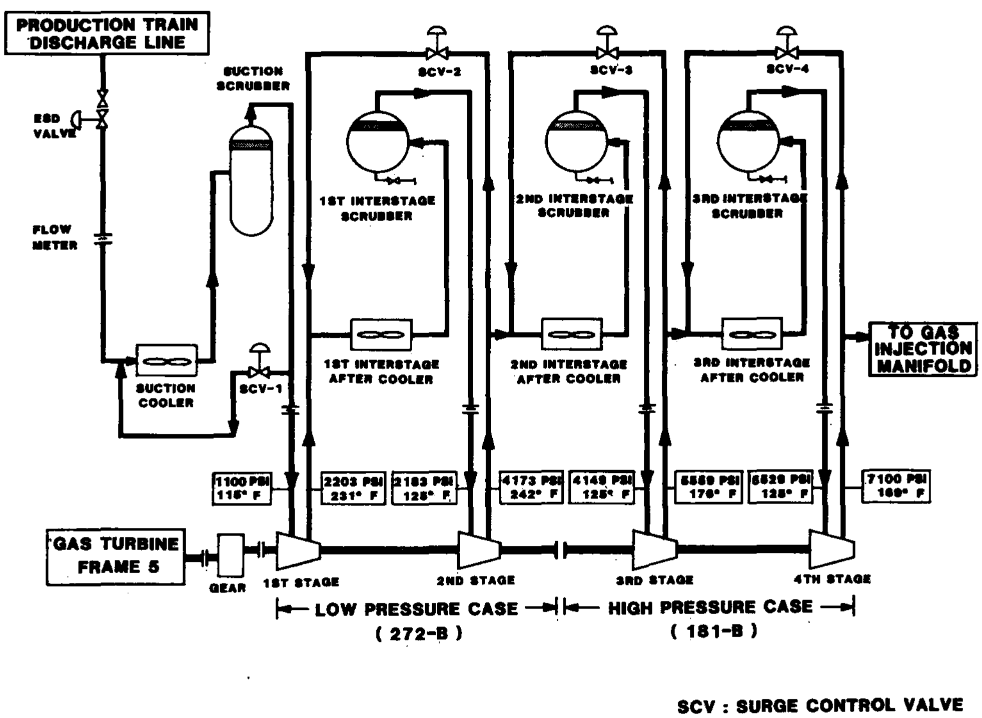

Fig. 2—Compressor flow diagram.

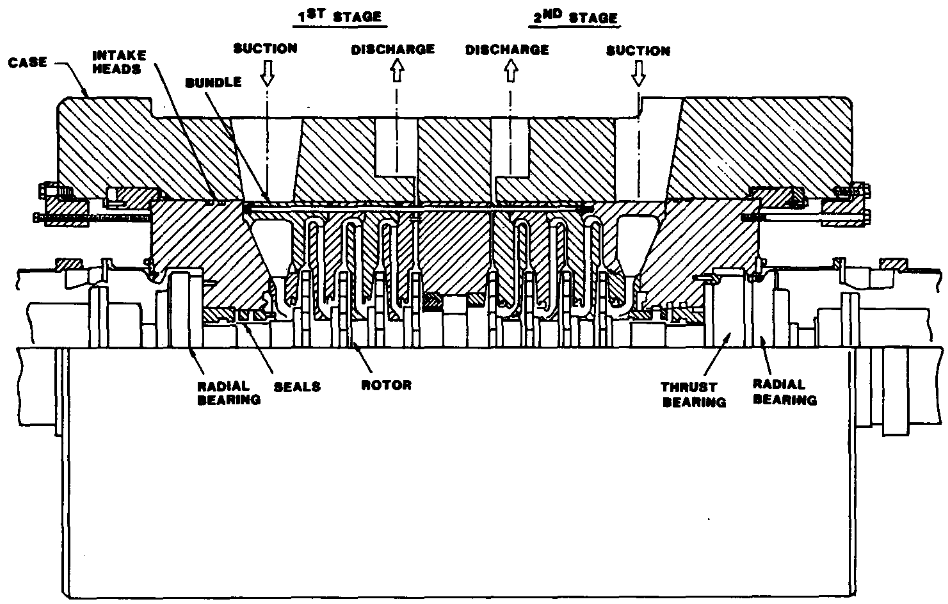

Fig. 3—Low-pressure 272B compressor.

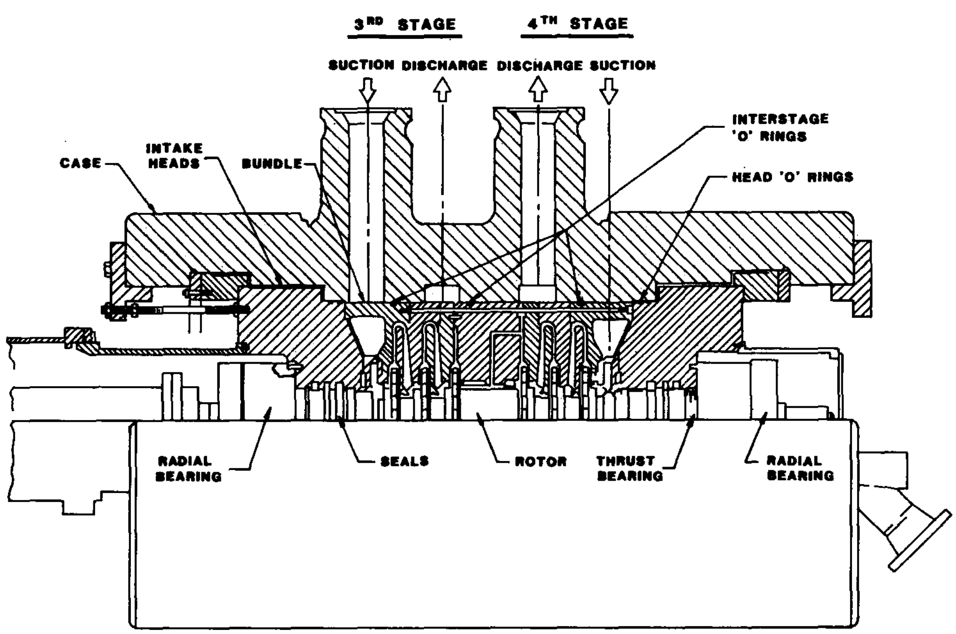

Fig. 4—High-pressure 181B compressor.

created by the high differential pressures across each stage.

Intake Heads. These are recessed into the compressor case and secured axially by shear and retaining rings that fit into annular grooves in the case bore. O-ring seals with backup rings form a pressure-tight seal between the outside of the intake heads and the case.

Inner Seals. Oil film seals are installed in the intake heads to prevent gas leakage into the bearing chambers. The seal oil system maintains a 5-psig [34.5-kPa] positive pressure across the seals and has a drain downline to seal oil traps that separate out the contaminated seal leakage gas from the seal oil.

Thrust Bearings. The thrust bearings have individually pivoted pads or shoes on either side of the thrust disk. The active shoes pivot or tilt against their supporting base ring to form an oil film "wedge" that protects the thinly babbitted faces from excess wear.

Journal Bearings. These are tilting pad type comprising a steel shell and five babbitt-faced pads or shoes, which tilt to form an oil film "wedge" in a manner similar to the journal bearings.

Vibration and Proximity Sensors. These are installed to detect and indicate shaft vibration and shaft axial movement.

Coupling. A lubricated coupling is used to connect the compressor low-pressure (LP) and high-pressure (HP) sections and the compressor LP section to the gear box.

Lube Oil and Seal Oil System. The lube oil and seal oil utilize a common reservoir with the suction to the seal

TABLE 1—SPECIFICATIONS FOR COMPRESSOR TRAINS

General	
Inlet pressure, psig [kPa]	1,100 [7584]
Inlet temperature, °F [°C]	115 [46.1]
Discharge pressure, psig [kPa]	7,100 [48 954]
Discharge temperature, °F [°C]	168 [75.6]
Design horsepower, bhp [kW]	25,200 [18 799]
Compressor speed, rev/min [rad/s]	10,080 [1056]
Design flow (100% speed), 10^6 scf/D [std m^3/d]	170 [4.81]
Weight flow, lbm/min [kg/min]	7,474 [3390]
Gas Properties	
Molecular weight	22.6
Main chemical components, %	
CH_4	75
CO_2	15
C_2H_6	5.5
C_3H_8	2.2
H_2S	trace

oil pump coming from the discharge of the lube oil pumps. Seal oil at a pressure of 5 psig [34.5 kPa] over the stage gas reference pressure provides positive sealing to the oil film type compressor seals. Three overhead seal oil tanks, on level control, maintain the required differential pressure across the seals. All reference lines are tied into the compressor first-stage suction line.

Both the lube oil and seal oil pumps are AC motor driven with 100% backup. Oil pressure to the bearings is regulated at 20 psig [138 kPa] from the lube oil pump discharge manifold, which operates at a pressure of 100 psig [690 kPa].

Gas Turbine Driver. ***The Turbine.*** The injection compressors are driven by a Frame 5, two-shaft, simple cycle, mechanical drive, gas turbine. The rated output is 28,000 bhp [20 888 kW] with 90°F [32.2°C] and 14.56-psia [1-MPa] absolute air inlet conditions and an exhaust temperature of 956°F [513°C].

The turbine has a 16-stage heavy-duty axial flow compressor driven at 5,100 rev/min [534 rad/s] by the HP shaft. The power, or LP, shaft is rated at 4,670 rev/min [489 rad/s] at 100% speed but can be varied from 90 to 105%.

Gear Box. A speed-increasing, single helical gear box drives the compressor from the turbine LP shaft. The gear ratio of 1:2.519 drives the two compressor sections in tandem at a design speed of 10,800 rev/min [1131 rad/s]. The turbine LP shaft is connected to the gear box by a flexible diaphragm coupling.

Fuel Gas. Train production gas is used for fueling the gas turbines. A fuel gas conditioning unit reduces the fuel gas pressure to 230 psig [1586 kPa] and reheats the gas in an indirect water bath heater to 120°F [48.9°C]. The gas then passes through an outlet scrubber and on to the gas turbine. The possibility of liquid carryover is minimized by the process of reheating and scrubbing.

Suction and Interstage Coolers and Scrubbers. ***Coolers.*** The four compressor stages each have an induced draft aerial cooler installed in the suction line.

The first-stage (suction) cooler reduces the temperature of the incoming gas to around 90°F [32.2°C]. This reduces compressor horsepower requirements and also reduces the inlet gas temperature when the compressor is on recycle.

The interstage coolers reduce the high gas discharge temperatures to limits acceptable for the compressor and also reduce the total horsepower developed by the compressor.

Scrubbers. Following each aerial cooler is a scrubber designed to knock out any liquid that may drop out during the cooling stage. The vertical suction scrubber has an automatic dump to a closed drain system. The spherical interstage scrubbers use hand-operated dump valves. All four scrubbers were manufactured from carbon steel and include stainless steel mist eliminators and grids.

Controls. Each of the four compressor stages is equipped with a recycle valve, which can be controlled by any of these controllers (Fig. 5): (1) compressor discharge pressure controller, (2) compressor suction pressure controller, (3) antisurge controller (one per stage), and (4) hand controller (to all stages).

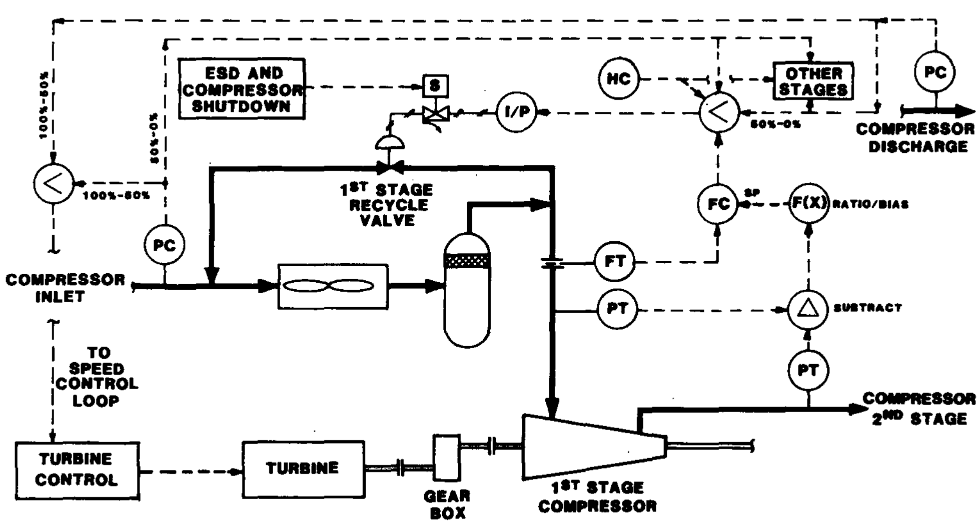

Fig. 5—Surge (first-stage) and pressure controls.

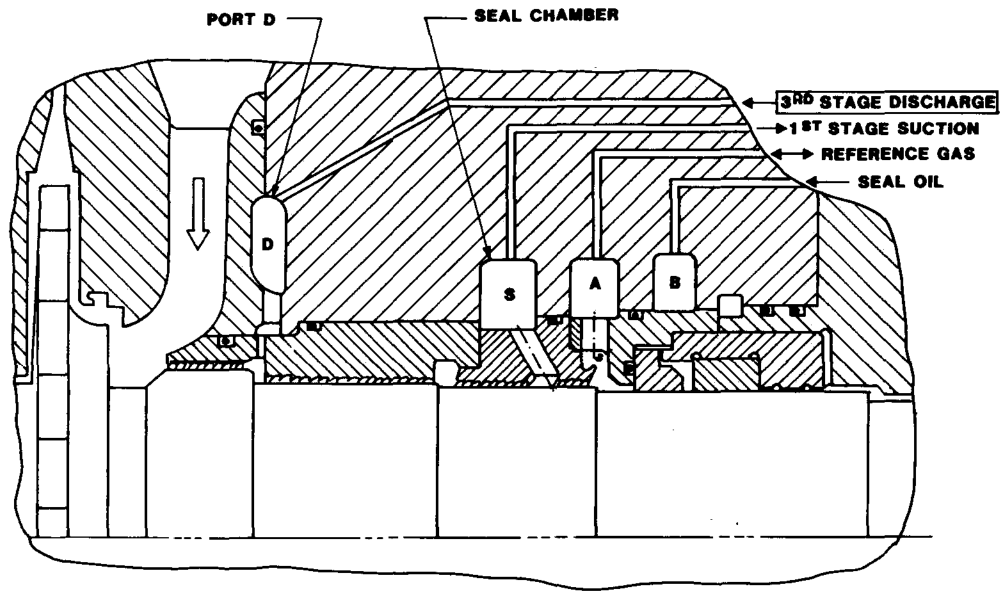

Fig. 6—181B fourth-stage labyrinth seal.

Output signals from each of the four controllers are fed to a low-signal selector. The lowest signal controls the respective stage recycle valve.

The first 50% of the suction pressure controller signal automatically reduces the compressor speed by way of the turbine control system. If the reduction in flow rate is insufficient to raise the suction pressure to normal, the second 50% of the control signal opens the surge control valves. This system is designed to ensure that gas pressure to the LNG plant is maintained at the expense of compressor throughput in the event of a plant upset.

A similar control regime is used for the discharge pressure controller to prevent overpressurizing the compressor discharge piping.

The set point of the interstage flow controllers is a function of the differential pressure across the stage and is based on the stage performance curves, the surge line, and the surge control margin.

If the interstage flow drops below the antisurge controller set point, the stage recycle valve opens to prevent the compressor going into minimum flow or surge conditions. The hand controller is used for loading and unloading the compressors during routine operation of the machines.

Six Years' Operating Experience

General Comments. The first compressor train (K301) was started in May 1977, followed by the second in Aug. 1977 (K351) and the third in Jan. 1978 (K201).

As could be expected, the high flow rates and high injection pressures, combined with the new compressor design concepts, led to many problems during the initial startup period. Some problems resulted in compressor shutdown while the fault condition was rectified. Other problems, while not requiring the train to be shut down, led to a rapid decline in compressor efficiency with a subsequent reduction in the gas injection flowrate.

This section describes the problems encountered during the startup phase and also the design changes required to maintain compressor injection efficiency over sustained operating periods.

Compressor Problems. ***Hydrate Formation.*** Shortly after startup, the differential pressure across the labyrinth seals on the third and fourth compressor stages began to fluctuate wildly and the pressure in the seal oil system increased to a level where the compressor had to be shut down.

After seal oil pressure recorders and labyrinth seal differential pressure recorders were installed to monitor the problem, we realized that the high pressure drop across the third-stage seals (3,100 psi [21 MPa]) and the fourth-stage seals (4,500 psi [31 MPa]) was creating a temperature drop sufficient to cause hydrate formation in the s-ports (Fig. 6). This was confirmed by reference to natural gas hydrate formation curves. This resulted in high reference gas pressures and caused gas to blow through the inner seal ring.

The immediate solution to the problem was to increase the suction temperature to third and fourth stages to 145°F [62.8°C] by controlling the louvres on the respective suction coolers. This temperature was the minimum temperature at which hydrate formation did not occur. This, however, was not a satisfactory long-term solution for two reasons: (1) compressor throughput was reduced (because of the higher suction temperatures), and (2) the elevated compressor discharge temperature caused expansion problems in the underground injection lines going to the GIW's.

The final solution was to pipe *uncooled* gas from the compressor second- and third-stage discharges to Port D of the third- and fourth-stage seals.

The high temperature of the discharge gas (about 190°F [87.8°C]) prevented hydrate formation across the seals and allowed the compressor to be run at original design conditions.

Interstage Leakage. An early problem encountered with the compressors was the decline in capacity that occurred

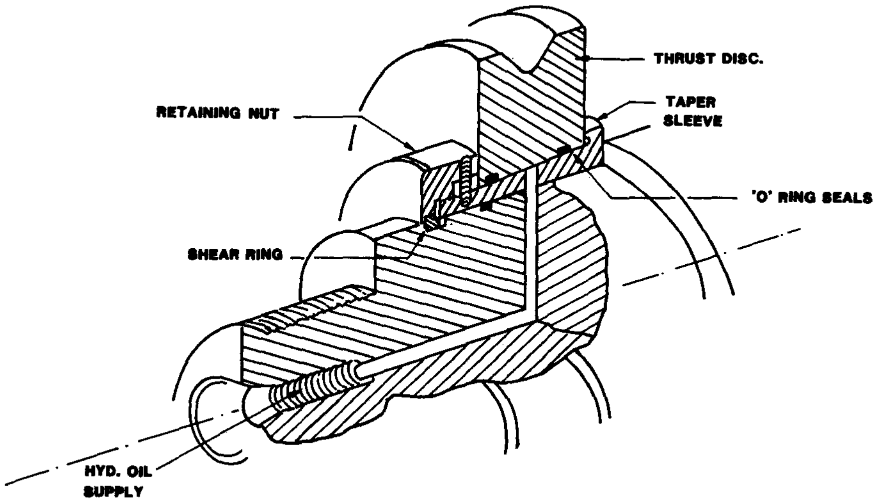

Fig. 7—Hydraulic fit thrust disk.

once the machine was in continuous operation. Even after only a few weeks, a reduction in throughput of between 5 and 10% was measured. Also, tests indicated that throughput decline increased as the number of compressor shutdowns and startups increased.

The problem was identified eventually as disintegration of the viton O-rings separating the first- and second-stage discharges on the LP barrel (Fig. 3) and the third- and fourth-stage discharges on the HP barrels (Fig. 4). The O-rings are located in grooves on the outside of the bundle and seal against the inside of the barrel.

After consultation with the manufacturer, it was decided that the 70-durometer O-ring material was becoming impregnated with gas while under pressure and that subsequent depressurization of the compressor led to swelling of the O-rings and their subsequent disintegration. The more frequently the O-rings were pressure cycled, the more quickly they deteriorated. The solution to the problem was to use 90-durometer fluorocarbon O-rings, which are less permeable to natural gas than the original 70-durometer material specified.

Also, before compressor reassembly, the O-ring mating surface on the inside of the compressor barrel was polished to prevent O-ring damage when the bundle was installed. Finally heavy-duty backup rings, with a tighter fit in the O-ring groove, also were specified.

These modifications considerably reduced the long-term decline in compressor capacity, and the interstage O-rings now show little sign of deterioration after having been in service.

A 63-root-mean-square (RMS) surface is now specified by the manufacturer for the barrel O-ring sealing surface instead of the 250-RMS surface originally specified.

Shaft Seal Area Erosion. During overhauls, deep circumferential grooving of the labyrinth area shaft sleeve was observed. The aluminum labyrinths, however, showed no wear or damage, and no evidence of rubbing between the shaft and labyrinth was detected. The increased clearance between the labyrinths and the shaft resulting from the grooving led to excessive seal gas leakage, with a subsequent decline in compressor efficiency and throughput.

To quantify the seal gas leakage, a 2-in. [5-cm] orifice plate was installed on the line taking seal gas from the compressor back to the first-stage suction. This confirmed that the seal gas flow was in excess of design.

The cause of the shaft grooving eventually was isolated to seal gas erosion, possibly resulting from drilling mud or reservoir particles being present in the gas (several new production wells had been put on line around the time that the problem occurred). Initially we thought that after the new wells had flowed for a time the gas would become cleaner and that the sleeve erosion would not recur on newly fitted sleeves. This, however, was not the case, and therefore we decided to change the shaft sleeves to Monel® with a hard facing 0.035 in. [0.089 cm] thick that had a Brinell hardness of 722 (the same hard facing had been used on the oil film sleeves with good results).

This proved a successful solution to the grooving problem and no evidence of shaft erosion or excessive seal gas leakage is now observed.

Thrust Disk Mounting. The thrust disk was designed originally to be a 0.000- to 0.001-in. [0.000- to 0.0025-cm] interference fit with the rotor shaft. However, accurately mounting the disk axially on the shaft to within the design tolerances required proved difficult. Also, because of fretting while in service, the disk and shaft corroded together, making the disk difficult to remove during overhauls.

The solution to both of these problems was to modify the shaft and use a hydraulically mounted thrust disk with a thin, tapered bushing between the disk and shaft (Fig. 7). This allowed precise axial positioning of the disk and, since the bushing is thin, when hydraulic pressure is released a tight interference fit is developed between both the disk and sleeve, and the sleeve and shaft. This provided a tight, accurately located fit between the thrust disk and shaft. Since the modification was incorporated, no fretting has occurred and the thrust disk is easy to fit and remove.

Head O-Ring Failures. Several forced compressor outages arose during the first year of operation because of leakage of the O-ring seals between the compressor heads and case of the high-pressure barrel (Fig. 4).

This was solved by changing the O-ring material to 90-durometer (as with the interstage seals) and by taking special precautions when assembling the heads into the barrel so that the O-rings were not damaged during assembly. A special jig was fabricated that enabled the heads to be aligned accurately with the case before assembly.

Turbine Problems. ***Turbine Burnout.*** In Feb. 1978, K351 turbine at Cluster 3 burned out. The exact cause of the failure was never resolved satisfactorily, but we suspected that a small slug of liquid condensate built up in the fuel gas line. This slug eventually was drawn into the turbine combustion chambers and caused a sudden and dramatic increase in the fuel Btu value. Once ignited, the turbine temperature protection system could not respond fast enough to prevent the extensive and costly damage that resulted from the ensuing thermal overload. To prevent a recurrence of this type, an additional fuel gas separator/filter unit was installed directly upstream of the turbine fuel control valves. The fuel gas piping was rerun also, eliminating all dead legs and low points where liquid could collect. The turbine fuel gas meter run was reorientated into a *vertical* position to prevent liquid buildup on either side of the orifice plate. Since then, no further problems of this nature have been experienced.

Turbine Exhaust Thermocouples. There are 12 exhaust thermocouples for the turbine temperature control loop plus 6 for the high-temperature alarm and shutdown system. These thermocouples burned out repeatedly, especially after a shutdown and restart of the machine. This resulted in the turbine operating on fewer thermocouples than was desirable to maintain a good average exhaust temperature signal to the temperature control loop and to the over-temperature and differential temperature alarm and trip protection systems.

Attempted solutions included insulating the thermocouple junction box, increasing the turbine exhaust compartment ventilation, and annealing the thermocouples before installation. The turbine manufacturer then developed a new, longer type of thermocouple designed to terminate *outside* the exhaust compartment, thus preventing overheating of the thermocouple terminations. These have increased the average life of the thermocouples to an extent, although the new design has not eliminated the problem completely.

Fire Protection System. A halon-based fire protection system is installed on each turbine. It is actuated by thermal detectors strategically placed around the three turbine compartments. Repeated failure of the detector system caused unnecessary compressor shutdown and halon waste. The problem was compounded by all the thermal detectors indicating on one common alarm in the control room, thus making it difficult to identify which detector had failed when an alarm occurred. This meant that the turbine had to be shut down, allowed to cool, and then each detector removed and tested in turn until the faulty one was located. This was time consuming and resulted in a substantial loss in revenue while the compressor was off-line.

Several shutdowns were caused also by intermittent failure of splices in the detector wiring resulting from high turbine compartment temperatures.

To rectify these faults, all the detector turbine wiring was renewed with new high-temperature fiberglass-insulated wiring, and a small semigraphic alarm unit was installed for each turbine showing *which* detector had failed when an alarm condition arose.

Instrumentation Problems. ***Interstage Flow Measurement.*** The suction flow to each of the four compressor stages originally was measured using a proprietary design flowmeter tube. These were specified because they are less expensive than many alternatives and have good pressure recovery characteristics. After startup, however, the measured flows did not agree with those predicted by calculation. Various attempts to identify and eliminate the inconsistencies were tried including rechecking and recalibrating instruments, checking the flow tube dimensions and location, testing of the field wiring, and rechecking the calculated flow, but still the error remained.

Attempts to check the basic meter factor were also difficult because of the unusual design of the tubes. In an attempt to isolate the problem, the meter tubes on one compressor were replaced with standard orifice plates. The measured flows then closely agreed with those predicted by calculation. Therefore, orifice plates also were installed on the other two compressors and the flow measurement error eliminated.

It should be noted that measured interstage flows are frequently more than the compressor suction (or compressor throughput) flow. This is because of the internal seal gas circulation that exists in labyrinth-seal-type compressors.

Also, the *actual* flow passing through a particular compressor stage is not necessarily the *measured* interstage flow. This is because not all seal gas circulation passes through the interstage meter. Alternatively, not all the gas measured by interstage flow meter actually goes to the compressor.

To demonstrate these points, the *measured* third interstage flow is 104.7% of the compressor suction flow, Q, whereas the *actual* third-stage compression flow is 102.7% of the compressor suction flow, Q.

Recycle Valve Vibration. Severe vibration of the third-stage 3-in. [7.62-cm] surge control valve and associated piping occurred when the compressor was on recycle (unloaded). The frequency and amplitude of the vibration was such that it sheared the 1.25-in. [3.18-cm]-diameter stud bolts between the valve and pipe flanges. To overcome this problem, a new 4-in. [10.2-cm] valve was installed with a 36-in. [91.4-cm]-long, 4×6-in. [10.2×15.2-cm] reducer on the valve discharge. This arrangement reduced the level of vibration considerably, and no further problem with the valve has been experienced.

Scrubber Level Switches. Each compressor suction scrubber is fitted with a displacer-type level switch designed to trip the compressor, should liquid condensate level build up above a fixed preset level. Intermittent level switch alarms and trips occurred, however, on all three compressor trains even though no liquid level was visible in the scrubber sight glass. It also was noticed that most of the erroneous alarms and trips were caused by the level switch on the two HP scrubbers. Inspection of the switches showed that some of the karbite displacers

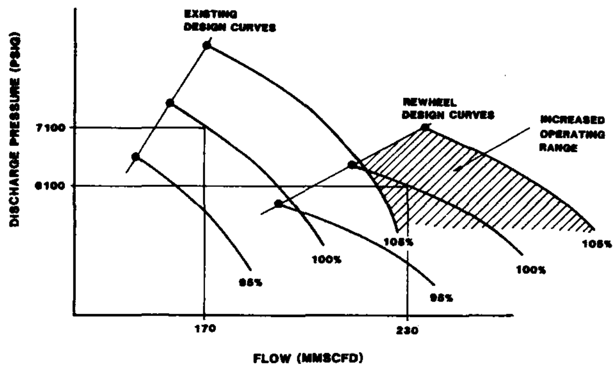

Fig. 8—Compressor performance curves.

had fractured, causing the displacer to become too light to counter the spring-balanced switch mechanism. The reason was gas impregnation into the displacers followed by expansion and cracking during pressurization and depressurization of the displacers during compressor operation. Also, because of the high pressure, the density of the gas in the fourth-stage scrubber was close to that of the liquid condensate, which the switches were designed to detect.

After consultation with the switch manufacturer, the problems were overcome by recalculating the weight and dimensions of the displacers and by treating the karbite with a sealant following manufacture.

Improving Performance

This section discusses some of the attempts to both monitor and improve compressor performance to optimize overhaul scheduling and improve compressor efficiency.

Performance Testing and Evaluation. To monitor compressor performance, monthly performance tests designed to determine the average rate of decline of throughput of the compressor and the polytropic efficiency of each stage are carried out on each compressor.

Before each test, calibrated standard pressure gauges and mercury-in-glass thermometers are installed at the suction and discharge to each compressor stage. The suction and interstage flow transmitters also are checked and calibrated. Tests are conducted normally at maximum speed to maintain maximum train throughput.

The test results then are fed into hand-programmable calculator and printer that calculates (1) compressibility factor at suction to each stage, (2) compressibility factor at discharge to each stage, (3) head developed per stage (ft·lbf/lbm), (4) mass flow rate per stage (lbm/min), (5) polytropic efficiency per stage, and (6) gas horsepower per stage.

The compressibility factors are based on pseudocritical pressure and temperature values for the gas, which are pre-entered into the calculator along with the gas molecular weight and gas specific heat at constant pressure.

Then the test points are plotted on the compressor design performance curves and referenced back to a constant head curve at 100% speed. This procedure enables direct comparison of the test results, since they are all based on the same reference conditions. The decline in polytropic efficiency and compressor capacity at reference conditions then can be monitored on a monthly basis. The decline in the total gas horsepower developed by the compressor indicates the overall decline in efficiency of the compressor train.

Mobil Oil Indonesia currently is developing an on-line performance monitoring program for inclusion in the field supervisory control and data acquisition (SCADA) system. This will enable performance tests to be performed quickly and at any time.

Although the results of these tests tend to be erratic (since *production* facilities like those at Arun are not designed specifically for *performance* evaluation) they do give good long-term efficiency trends that can be used for performance comparisons and maintenance planning.

Compressor Rewheeling. As already stated, the original design of the injection compressors at 100% speed was 170×10^6 scf/D [4.81 std m^3/d] with 1,100-psig [7584-kPa] suction pressure and 7,100-psig [49-MPa] discharge pressure. This design was based on the reservoir pressure in 1977 (Fig. 8).

By 1984, the compressor discharge pressure had dropped to 6,100 psig [42 MPa] and injection rates increased as predicted by the compressor performance curves.

However, because the compressor operating point was moving increasingly away from the original design point, the polytropic efficiency of each state was dropping and stone wall conditions were being approached. This meant that the compressors were operating inefficiently, turbine horsepower was being wasted, and potential additional gas injection was being lost.

To utilize the available turbine horsepower efficiently, a new rotor and bundle assembly was designed based on a discharge pressure of 6,100 psig [42 MPa]. This required designing new impellers for both shafts and reducing the number of impellers on the low-pressure shaft from four to three per stage. The new bundle/rotor assembly was designed to fit inside the existing compressor cases, thus reducing the cost of the modification and obviating the need for any piping changes.

The first new rotor and bundle assembly was installed in early 1984 and gas injection rates increased by 35×10^6 scf/D [0.99 std m^3/d] to about 255×10^6 scf/D [7.22 std m^3/d]. It is expected that the compressors will be rewheeled several more times during the projected life of the injection trains. This will optimize the use of available turbine horsepower and produce maximum compressor throughput.

Optimization of Operating Conditions. To maximize injection throughput, the compressor suction pressure (which, apart from line pressure drops, is essentially the same as the production train separator pressure) is kept at the maximum allowable pressure under the codes to which the piping was designed.

The temperature of the inlet gas is controlled by the suction coolers and is maintained at the minimum temperature that will not produce any liquid at the compressor inlet *or* cause a recurrence of the hydrate problem previously discussed. The compressors also are run at the maximum obtainable turbine speed, which is normally around 101% of the design speed. The turbines run continuously on exhaust temperature control.

The combined effect of increased suction pressure, decreased discharge pressure, increased speed, minimum gas inlet temperature, and rewheeling has resulted in the compressor capacity being increased from the design figure of 170×10^6 scf/D [4.81 std m^3/d] to a present throughput of 255×10^6 scf/D [7.22 std m^3/d] per machine.

Maintenance Scheduling. An economic evaluation has been made of compressor performance to determine both the optimal frequency of overhauls and to determine how the compressor and turbine overhauls could be best integrated together.

The results of the study were used to determine the point at which the *loss* in compressor throughput from maintenance downtime for overhaul became justified by the *gain* in compressor throughput as a result of that overhaul. It was found that several millions of dollars per year could be lost as a result of nonoptimal overhaul scheduling.

From the monthly performance tests carried out on the compressors, the average decline in capacity was found to be approximately 2.1×10^6 scf/D [0.059 std m^3/d] gas per compressor per month. This decline then was equated to lost revenue from reduced condensate production. By analyzing lost production revenue during overhaul and the total cost of the overhaul, we determined that a 13-month overhaul cycle of the compressors was economically optimal.

Conclusions

The design, commissioning, and operation of the gas reinjection facilities at the Arun field have been successful.

The many problems that inevitably beset any such project incorporating recently developed and untried technology all have been resolved, and the units now run almost continuously. The throughput efficiency is maintained at an acceptable level for extended periods, and maintenance downtime has been reduced to a minimum.

The lessons learned at Arun regarding the use of high-pressure, high-volume, natural gas reinjection facilities are summarized in the following recommendations.

1. Check for the possibility of hydrate formation across the labyrinth seals.
2. Use 90-durometer fluorocarbon O-rings.
3. Hard face the rotor sleeves under all labyrinths.
4. Incorporate hydraulically fitted thrust disks.
5. Avoid dead legs and low points when designing fuel gas line piping for gas turbines.
6. Use high-temperature insulation on all turbine instrumentation wiring. Do not use splices.
7. Be cautious if selecting proprietary design flow measurement elements.
8. If possible, check vibration characteristics of recycle valve piping and ensure that valves are sized correctly.
9. Be careful in the selection of material used for level switch displacers.
10. Conduct regular performance tests of the compressor to check compression efficiency.
11. Consider a rewheeling program when compressors are used for gas reinjection purposes.
12. Carefully review maintenance scheduling and procedures to optimize compressor efficiency and utilization.

SI Metric Conversion Factors

°F	(°F−32)/1.8		= °C
psi	× 6.894 757	E+00	= kPa

*Conversion factor is exact.

JPT

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